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Transverse Vibrate Analysis and Optimization for a Tuck Cab

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Abstract

Built vehicle vibrate system modal of a truck and performed modal analysis of every sub-system. The result from modal analysis and excitation frequency showed that the main reason that cause transverse vibrate of the truck under the velocity of 65km/h was as follows: the excitation from wheel unbalance in this velocity closes to the first-order natural frequency of the frame sub-system. The experiment result showed that the PSD of the cab floor reduce of 76.7% and the ride comfort of the truck is better after improved in the frame and wheels balanced.

Keywords: Cab; Transverse vibrate; Modal analysis; Self-power spectrum.

INTRODUCTION

Due to different natural frequency of the automobile parts, the strong vibration of the vehicle or part can affect the safety reliability and ride comfort of the car, and even lead to the destruction of the structure, affecting the service life. The automobile external and internal shocks is often caused by the uneven speed, movement direction, the imbalance between the wheel and transmission system and the shock of the gears and so on [1-2].

Research shows that the frame can produce resonance leading to the cab transverse vibration when the dynamic load is very big and has the road random vibration loading[3]. In order to suppress the resonance, the excitation frequency of the excitation source of the engine wheel can be controlled, also can by adjusting the structure of certain components to change natural frequency, thus reduce the transverse vibration of the cab.

Usually the vibration problem of automobile is solved by calculating the modal of each part in the car[4]. But in the process of vehicle running, the assembly components related interactions. therefore, the resonance problem of automobile is solved by analyzing the vibration frequency. The parameters of the natural frequencies and the natural modal for the structural system can be obtained by modal parameter identification. The experimental modal analysis and finite element modal analysis[5-6] are the effective means to identify the dynamic performance of automobile structure and them have been widely used in the research of dynamic performance of automobiles.

Aiming at the abnormal vibration problem of a certain type of truck, the author of this paper, established a vehicle vibration system model firstly. then based on the modal analysis of frame system and the cabin system, modal frequency and vibration of each order were obtained for frame system and the cabin system. And then the main cause of abnormal vibration was found combined with the analysis of the vehicle drive source. In the end, the improving measures were put forward, and the results were improved through test verification.

MODE OF VEHICLE VIBRATION SYSTEM

The test truck is mainly made of the main auxiliary frame and the container with bolts. The power assembly has a higher mode than the frame, and the rubber damper of frame connection is very hard. So the main auxiliary frame, container assembly, power assembly and so on as a system, namely the frame system. Between the bridge and the main frame is connected with a rubber damper. Because of the small hardness of the rubber damper, the cab is a separate system, called the bridge cab system. The main frame and the axle are connected by a suspension bridge composed of a steel plate spring and a damped damper. The automobile vibration system model is set up as shown in figure 1. To avoid resonance, the free modal frequency of the system should avoid the excitation source frequency. Therefore, the free modal analysis is carried out on the frame system and the cab system.

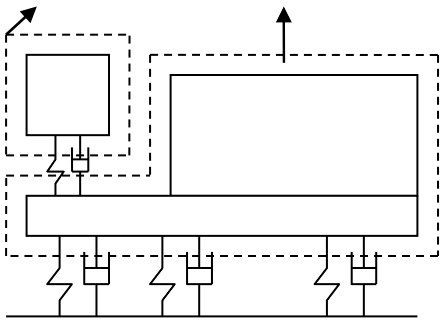

Figure 1. Vehicle vibrate system modal

For an n-rank vibration system, its motion equation is：

$$[M]\{\ddot{x}\}+[C]\{\dot{x}\}+[K]\{x\}=\{f\} \qquad (1)$$

General Training Module，[M]、[C]、[K]、{f}and{x}are respectively represent n order mass matrix, stiffness matrix, force vector and response vector. Taking the Laplace transform of it:

$$([M]s^2+[C]s+[K])\{X(s)\}=[Z(s)]\{X(s)\}=\{F(s)\} \quad (2)$$

There into:

$$X(s)=\int_{-\infty}^{+\infty}X(t)e^{st}dt \quad (3)$$

$$F(s)=\int_{-\infty}^{+\infty}F(t)e^{st}dt \quad (4)$$

$$Z(s)=[M]s^2+[C]s+[K] \quad (5)$$

The above equation reflects the dynamic characteristics of the system. It's called the system dynamic matrix. The inverse matrix is:

$$H(s)=[Z(s)]^{-1}=\frac{adj([Z(s)])}{|[Z(s)]|} \quad (6)$$

The matrix $H(s)$ is the transfer function matrix for the system. There into, the $adj[Z(s)]$ is adjoint matrix of the $[Z(s)]$, $|Z(s)|$ is determinant of the $[Z(s)]$.

By the characteristic equation $|Z(s)|=0$,We can get the roots of the system characteristic equation that is the eigenvalues of the system. They determine the resonance frequency of the system. Each eigenvalue λ corresponds to an eigenvector $\{\phi\}$. These guys are orthogonal. There is the following relation:

$$\{\phi_q\}^T[M]\{\phi_r\}=\begin{cases}0, r\neq q\\ m_r, r=q\end{cases}$$

$$\{\phi_q\}^T[K]\{\phi_r\}=\begin{cases}0, r\neq q\\ k_r, r=q\end{cases} \quad (7)$$

General Training Module, $\{\phi_q\}$ and $\{\phi_r\}$ are respectively represent the q and r order eigenvector; m_r and k_r are respectively represent the r order modal mass and modal stiffness. I'm going to take n eigenvectors ϕ_i order $n\times n$ order matrix divided by column. This matrix is a vibration matrix Φ and modal coordinates Q:

$$\Phi=[\phi_1 \quad \phi_2 \quad \phi_3 \quad ... \quad \phi_n] \quad (8)$$

$$Q=\{q_1(\omega) \quad q_2(\omega) \quad q_3(\omega) \quad ... \quad q_n(\omega)\}^T \quad (9)$$

The vibration matrix $[\phi]$ can decouple the modal mass and modal damping of the coupling together, and then the modal parameters of each order mode are obtained[7].

MODAL ANALYSIS OF VEHICLE SYSTEMS

The experimental modal system is composed of signal measurement and data acquisition system and signal analysis system. In this experiment, the excitation hammer was used for single point excitation. An excitation hammer can generate transient excitation signals. The order incentives depend on the measuring point number, and the force window is to attenuate the noise signal. The response signal was measured by ICP acceleration sensor, and the hanning window was used to reduce the energy leakage. DH5920 was used for signal acquisition and data analysis. In order to reduce the computation amount of the computer and obtain the first order mode of the cab, the frequency of analysis is defined as 120Hz. The sampling frequency is usually 2.5~3 times of the analysis frequency, so it is set to 400HZ. In order to improve the test precision and reduce the random error, the measured frequency response function is processed on average[8].

Frame system modal test is conducted on get rid of the cab of the vehicle, according to the position of excitation response points selection principle[9], the arrangement of measuring points on the longitudinal beam hinged with engine transmission, such as suspension parts ,longitudinal beam ,first beam and key part of a beam. The modal analysis of the test signal is obtained, and the first five order modal frequency table of the frame is shown in table 1. The first-order mode vibration mode is shown in figure 2, which is the upper and lower bending mode.

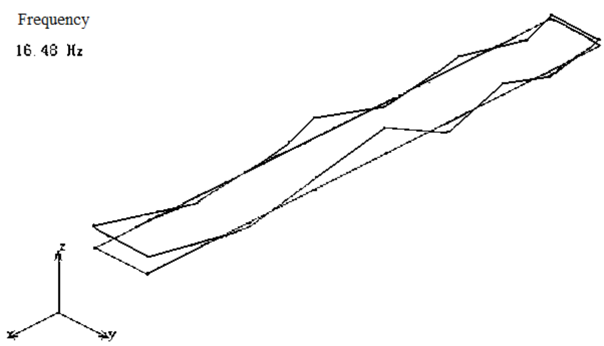

Figure 2. The first test modal shape of the frame system(restricted)

A finite element model based on solid unit solid45 is established. The engine gearbox auxiliary frame and container are uniformly applied in the corresponding position in the form of uniform quality points. The fuel tank and battery are applied to the frame in the form of concentrated mass points. The entire frame has 127,551 nodes, 77610 units and 1320 quality point units. The connection between the three suspension and the frame of the car is one end of the hinge and one end of the chute. Therefore, it is necessary to add three displacement constraints on the hinged end. The displacement constraint of the front and back direction of the frame is only released at the end of the slide. In order to avoid the over-static frame of the frame, only three direction displacement constraints are added at the front end of the hinge. Longitudinal and left-lateral displacement constraints are applied on the other two hinged branches. Using the Lanczos method[10] for finite element calculation, the natural frequency of each order of the frame system is shown in table 1.

Table 1. The test and FEA nature frequency of the frame system(restricted)

Order	Test frequency	Calculated frequency	Error(%)
1	16.5 Hz	15.8 Hz	4.24
2	20.8 Hz	21.2 Hz	1.92
3	29.3 Hz	29.5 Hz	0.68
4	34.8 Hz	35.7 Hz	2.59
5	39.7 Hz	40.3 Hz	1.51

It can be seen from table 2 that the calculation of frame system and experimental mode frequency error are within 5%. First mode as shown in figure 3, it has up and down bending, and mode fits in with the experiment showing that the established finite element model of frame system is reasonable. Free modal analysis can be carried out according to the frame system model.

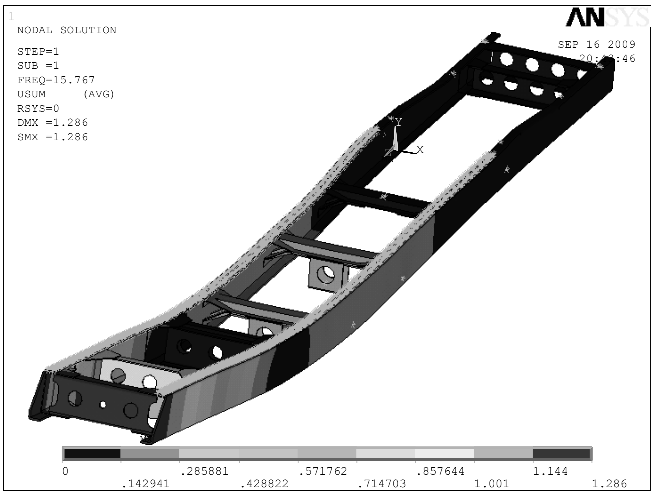

Figure 3. The first FEA modal shape of the frame system(restricted)

After the constraint was removed, the natural frequencies of the free modes of each order were calculated as shown in table 2, and the first-order mode was shown in figure 4, which was the torsional vibration of the left and right sides, and the torsion was larger at the front end of the frame.

Table 2. The FEA nature frequency of the frame system(free)

Order	1	2	3	4	5
Frequency(Hz)	4.80	13.13	13.71	19.35	27.82

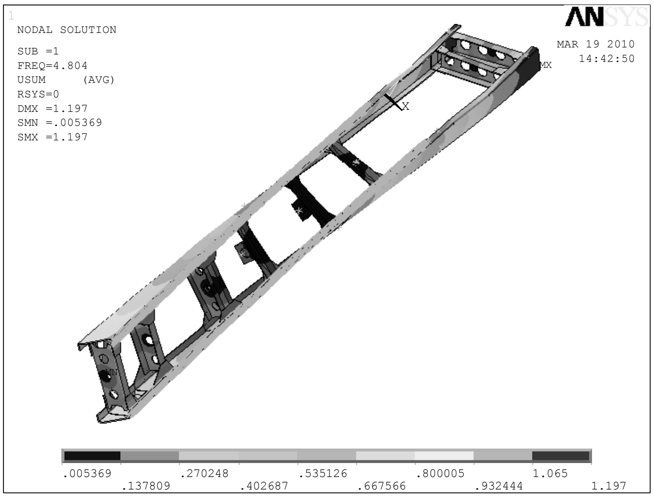

Figure 4. The first FEA modal shape of the frame system(free)

To make the cab in a free state, the cab should be placed on the soft elastic support or suspension by the elastic rope[11], and this test choose the former method in the cab. Dividing the grid in the cab and arranging the measuring points. and measuring point distribute in the cabin outside, door or bridge floor. There are total of 423 points, as shown in figure 5.

Figure 5. The measure dots of cab system's modal test

The test system and the frame test condition are the same. Through the modal analysis on the test signal, we get the inherent frequency of the bridge as shown in table 3. The first mode is the whole torsion mode, as shown in figure 6.

Table 3. The test nature frequency of the cab system

Order	1	2	3	4	5
Test frequency(Hz)	17.0	34.8	41.5	52.5	54.3

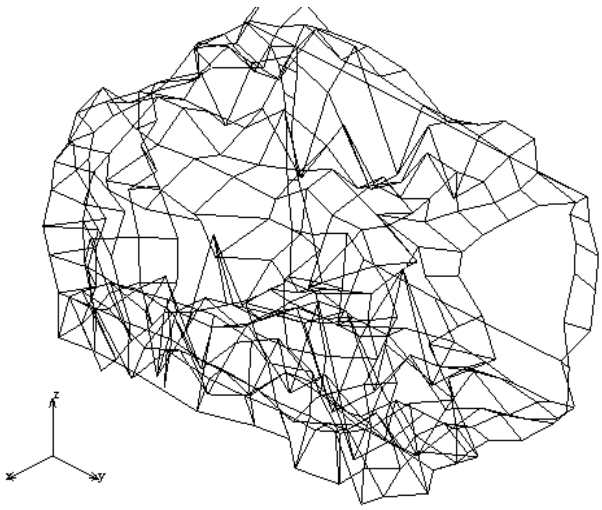

Figure 6. The first modal shape of the cab system

RESULTS ANALYSIS AND IMPROVEMENT

The driving force is divided into three types: first random excitation of the road surface roughness to the secondly, the simple harmonic excitation caused l pressure of working stroke and reciprocating inertia fo piston when the engine is running; thirdly, the harmonic excitation caused by the dynamic imbalance wheel. The engine of the test car is a four-cylinder four engine. The test speed is 65 km/h, and the four gears are running at a constant speed. At this point the engine speed is 1850r/min, and the speed ratio is 1. Therefore, the frequency of the main excitation source in this working condition is shown in table 4:

Table 4. The primary excitation frequency of the vehicle

Excitation source	Engine	Gearbox	Wheel	Road
Frequency(Hz)	61.66	30.83	4.88	0~20

The table 4 shows that the excitation frequency of the engine and transmission is high frequency, and the road excitation is random excitation. Only the first order torsion frequency of the wheel dynamic unbalance excitation and the free state of the frame system is close. Considering that the frame system is supported by the steel plate spring, it is impossible to be bound by the three-way displacement while working in a kind of approximate free state. Therefore, it is reasonable to think that the wheel dynamic imbalance excitation causes the torsional resonance of the frame.

On the other hand, the modal frequency of the cab is much higher than the first order mode frequency of the frame system, and it is only possible to follow the frame sway without causing resonance. Judging from this, the wheel dynamic imbalance excitation leads to lateral vibration of the cab.

The vibration control mainly starts from the two aspects of excitation and response, Thus, the method to solve the problem is obtained. One is to reduce the vibration of the excitation source, which is to reduce the dynamic balance of the wheel. The second method is prevent resonance. By improving the structure of the frame, the first order natural frequency of the frame is changed, and the frequency of dynamic imbalance of the wheel is avoided.

It can be seen from figure 4 that the first order free mode front vibration amplitude of the frame system is the largest. In order to change the natural frequency of frame, the front end rigidity of frame should be increased. Three kinds of improvement schemes are proposed for the frame structure: the one scheme is adding reinforcement to the first beam of the frame; the second scheme is adding a crossbeam to the second beam of the frame; the third scheme is increasing the thickness of the first beam of the frame by 3mm, as shown in figure 7.

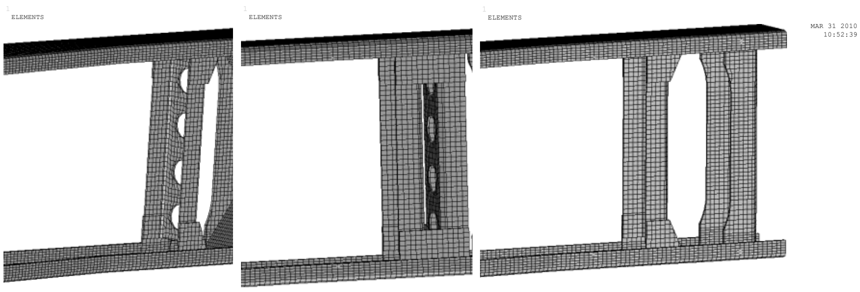

Figure 7. The improved frame structure

The first order natural frequency of the improved frame system is 4.76Hz 5.13Hz and 4.82Hz respectively, and the vibration mode change is not big. It is show that the method of reducing vibration alone from the response path is not ideal. Therefore, two measures are considered to reduce the torsional vibration of the frame:

1) Add a crossbeam to the second beam of the frame.
2) The uneven measurement of the wheel is adjusted to reduce the vibration effect of the excitation source.

In order to verify the improved effect, learning about the improved vibration characteristics of the truck before improvement, so two road test experiments were conducted on a certain b-grade road section. The acceleration sensor is arranged on the floor of the cab, measuring the acceleration signal of a truck at a constant speed 65km/h. Its self-power spectrum(SPS) is shown in figure 8 and figure 9. You can see from the picture, the improved under the main frequency power spectrum is reduced, which was 76.7% lower in the main impact frequency of 4.88 Hz. The mean square root value of acceleration before and after improvement was reduced from 2.3321m/s2 to 1.4792m/s2. The sway of the ride in the cab was significantly reduced, and the comfort got a very big improvement.

CONCLUSION

The following conclusions are drawn from the above analysis:

a) The experimental results show that it is feasible to carry out the modal analysis of the subsystems in the vehicle vibration system respectively.

b) The modal calculation results in the constraint state of the frame system are consistent with the test results. This demonstrates that the modeling method and analysis method are feasible. It provides the basis for the vibration analysis of the vehicle.

c) The free modal analysis of the frame system shows that the natural frequency of the first order torsion mode of the frame is 4.80Hz, which is close to the wheel frequency (speed 65km/h), resulting in lateral vibration of the cab.

d) Road test results show that the self-power spectrum of the vibration acceleration signal of the bottom plate of the cab is reduced by 76.7 after the improvement of frame structure and wheel dynamic balance. The sway of the ride in the cab was significantly reduced, and the comfort got a very big improvement.

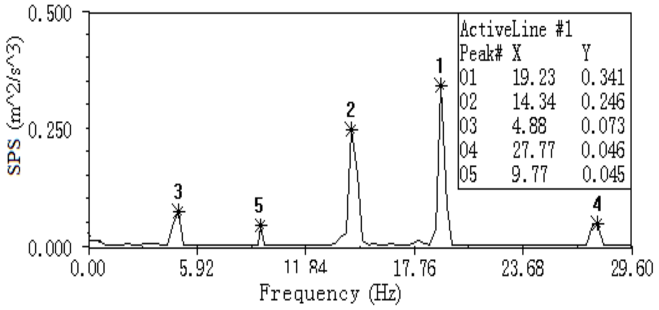

Figure 8. The PSD of the cab floor’s vibrate signal originally

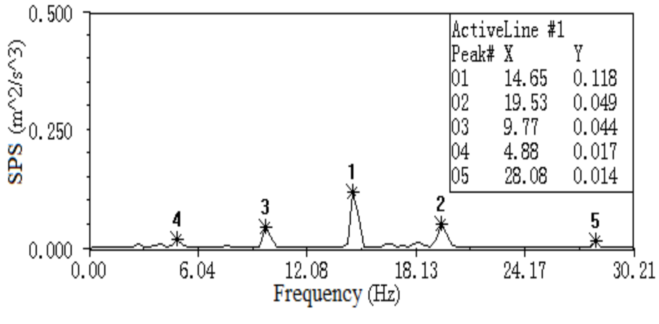

Figure 9. The PSD of the cab floor’s vibrate signal improved

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